An LPV/$\mathcal{H}_\infty$ Integrated Vehicle Dynamic Controller

Soheib Fergani, Olivier Sename, and Luc Dugard

Abstract—**This paper proposes a new multivariable linear parameter varying (LPV)/$\mathcal{H}_\infty$ control strategy for global chassis control. The main objective is to handle critical driving situations by activating several subsystems (semi-active suspensions, active steering, and electromechanical braking actuators) in a hierarchical way. The main idea is to schedule the three control actions (braking, steering, and suspension) according to the driving situation evaluated by a specific monitor. Indeed, on one hand, rear braking and front steering are used to enhance the vehicle yaw stability and lateral dynamics, and on the other hand, the semi-active suspensions are used to improve comfort and car handling performance. Due to the LPV/$\mathcal{H}_\infty$ framework, this new approach allows a smooth coordination to be reached between the various actuators, to ensure robustness and stability of the proposed solution, and to significantly improve the vehicle dynamical behavior. Simulations have been performed on a complex full vehicle model, which has been validated using data obtained from experimental tests on a real Renault Mégane Coupé. Moreover, the suspension system uses magnetorheological dampers whose characteristics have been obtained through experimental identification tests. A comparison between the proposed LPV/$\mathcal{H}_\infty$ control strategy and a classical linear time-invariant/$\mathcal{H}_\infty$ controller is performed using the same simulation scenarios and confirms the effectiveness of this approach.**

Index Terms—**Braking, linear parameter varying (LPV), semi-active suspension, steering, vehicle dynamic control (VDC), $\mathcal{H}_\infty$.**

I. INTRODUCTION

ROAD safety has been an international stake over the past few decades. In particular, enhancing driving characteristics by ensuring stability in critical situations (i.e., safer vehicles) has been recently the main issue for both academic and industrial communities. A new trend is to develop multivariable vehicle dynamic control (VDC) strategies involving several actuators that collaborate to enhance the vehicle dynamics. Such a multiple-input–multiple-output controller allows to adapt the vehicle behavior to the driving situations. This paper focuses on improving both comfort and safety by coordinating the use of the braking, steering, and semi-active suspension subsystems.

In the last decades, several studies dealing with the control of automotive dynamics have been proposed, which are based on single-input–single-output control solutions as in [1], where only the braking control is used to improve the lateral and yaw behaviors of the vehicle. Other studies have been concerned with the improvement of the lateral dynamics behavior using braking and steering actuators, as in [2], by using an optimal nonlinear vehicle control. Some of the authors' results in [3]–[5] present scheduling policies of braking /steering actuators, which is a step toward a full coordinated control. However, it has been proven that steering and braking actuators are not sufficient to improve all the vehicle dynamical behaviors, such as the roll characteristics, which can be handled through controlled suspension systems. Indeed, the suspension system plays a key role in the automotive vertical dynamical attitude. Most of the control strategies developed for such systems allow to achieve the performance objectives concerning the passenger comfort and the car roadholding, as presented in [6]–[9].

The study of each of the suspension, steering, and braking systems has shown that an independent design for each one of them may lead to performance conflicts due to the different interactions between the vehicle dynamics. A solution to this problem is to consider integrated control strategies that allow managing multiobjective performances through all the available actuators and sensors. The interest for this type of vehicle control has increased in several academic and industrial research centers. As a result of the interactions between the vertical and lateral dynamics, as aforementioned, new vehicle control methodologies, including suspension and braking or steering actuators, have been developed, e.g., a hierarchical fuzzy-neural control of an antilock braking system and active suspension in [10].

As an illustration of integrated VDC solutions, the authors have developed a new robust control structure to enhance the overall vehicle dynamics using a coordinated approach of the steering, braking, and magnetorheological semi-active dampers suspension systems. A robust linear parameter varying (LPV)/$\mathcal{H}_\infty$ based on LMI's resolution in the LPV framework for those subsystems is developed in [11]. Moreover, some first results concerning the robust multivariable control using the three types of actuators are established and validated in [12]. In this paper, a new LPV/H_∞ control strategy (see Fig. 1) provides a hierarchical collaborative coordination between the actuators of semi-active suspension, steering, and braking subsystems to enhance the vehicle dynamics and prevents conflicts in terms of performance objectives. On one hand, this VDC strategy combines the monitoring of the driving situation and the corresponding coordination of the actuators; on the other hand, the LPV/H_∞ frame allows a smooth and flexible use of the actuators, with adaptation to the driving situation, while guaranteeing the robustness of the proposed control. The designing of controllers focuses on enhancing the overall

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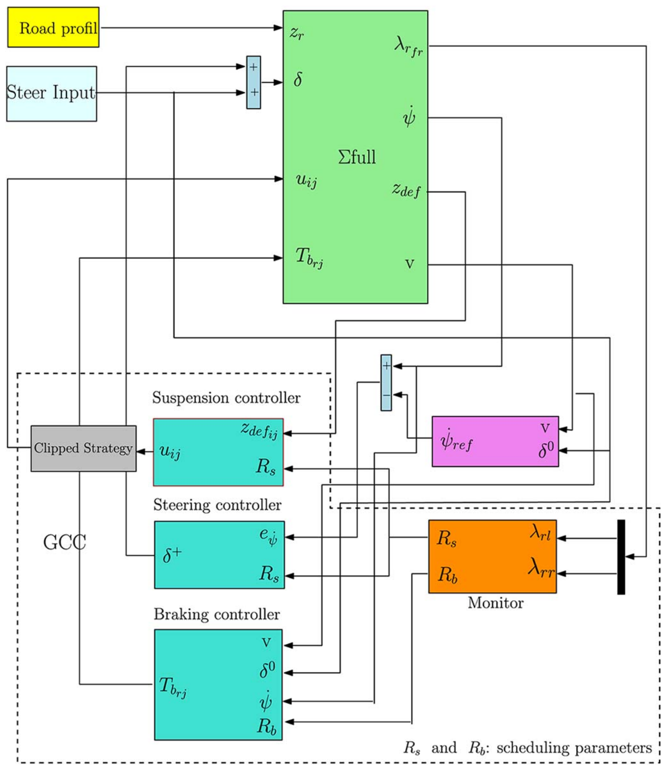

Fig. 1. GCC implementation scheme.

vehicle dynamics, namely, vertical, lateral, and longitudinal dynamics.

Moreover, in this paper, semi-active suspensions are considered (while in [13], active systems were used), which suits better industrial requirements (energy saving). This strategy is adapted to aid the driver, depending on how dangerous the situation is. First, it selects the best actuator coordination to avoid accidents, and second, it limits the unnecessary use of the actuators to save energy.

More precisely, this paper presents, in a unified and detailed way, some results of previous studies and enhances those studies by adding the main following contributions.

- The use of the real data input collected on a real car (Renault Mégane Coupé) running on a track to evaluate the effectiveness of the proposed LPV/$\mathcal{H}_\infty$ strategy in a more realistic simulation framework;
- The use of the semi-active magnetorheological dampers (MRDs) for the suspension control implementation. Such dampers allow the achievement of good comfort, enhancement of the roadholding, and keeping a safety suspension deflection.

This strategy is summarized in the following implementation scheme (see Fig. 1), including the vehicle's model, the monitoring approach, and the subsystem controllers. This paper is organized as follows. Section II presents the overview of the main contribution of this paper, which is the coordination between semi-active suspension, steering, and braking actuators, and the synthesis of various controllers to enhance vehicle performances and attitude. The control syntheses for the braking/steering subsystem (respectively, suspension subsystem) are detailed in Sections III (respectively, IV). In Section V, time-domain simulations are performed on a complex nonlinear full vehicle model equipped with semi-active suspension MRD. It also emphasizes the contribution of the proposed LPV strategy by comparing it to the linear time-invariant (LTI) strategy. Conclusions end this paper.

Paper Notations: For modeling and simulation purposes, the following vehicle parameters and notations are adopted. Throughout this paper, indexes $i = \{f, r\}$ and $j = \{l, r\}$ are used to identify the vehicle's front and rear, and left and right positions, respectively. Moreover, since the full vehicle model is used, some notations will appear. Index $\{s, t\}$ holds for suspension and tire forces, respectively. $\{x, y, z\}$ holds for forces and dynamics in the longitudinal, lateral, and vertical axes, respectively. Then, let $v = \sqrt{v_x^2 + v_y^2}$ denote the vehicle speed, $R_{ij} = R - (z_{us_{ij}} - z_{r_{ij}})$ the effective tire radius, $m = m_s + m_{us_{fl}} + m_{us_{fr}} + m_{us_{rl}} + m_{us_{rr}}$ the total vehicle mass, $\delta = \delta_d + \delta^+$ the steering angle (δ_d is the driver steering input, and δ^+ is the additional steering angle provided by steering actuator), and $T_{b_{ij}}$ the braking torque provided by the braking actuator (see Section II).

II. New Global Chassis Control Strategy: Supervision and Synthesis

Here, the main contribution of this paper, namely, the multivariable VDC strategy involving front active steering, rear braking, and semi-active suspension (see Fig. 1), is presented. The main idea is to synthesize two controllers, one dedicated to the lateral dynamics and the other to the vertical dynamics, which will be coordinated due to the scheduling parameters R_b (braking) and R_s (suspension and steering).

A. Driving Situation Monitoring

The monitoring of the driving situation has been selected following [13] from the longitudinal slip ratio of the rear wheels (s_{ij}) since it considerably affects the yaw stability and the car handling attitude.

- **Braking monitor**

$$R_b = \min_{j=l,r}(r_{b_j}) \tag{1}$$

is a function of the absolute value of the slip ratio ($|s_{rj}|$). r_{b_j} is defined as a relay (hysteresis like) function: $\rightarrow 0$ when "on", $\rightarrow 1$ when "off". The switch "on" (respectively, "off") threshold is s^+ (respectively, s^-). When the slipping is low, the vehicle is in a normal situation; hence, $R_b \rightarrow 1$. When the slip ratio rises and becomes greater than s^+, a critical situation is detected; then, $R_b \rightarrow 0$. Since R_b is function of the slip ratio, s^+ and s^- are chosen according to the tire friction curve. Here (and in a general case), $s^+ = 9\%$ and $s^- = 8\%$, in order to delimit the linear and peak tire friction force with the unstable part of the tire (see [14]).

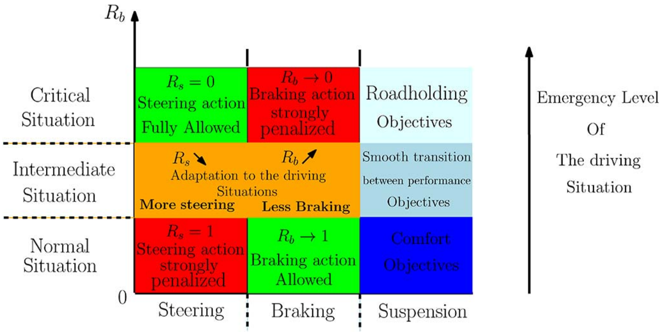
R_b
Critical Situation
Intermediate Situation
Normal Situation
0
$R_s = 0$ Steering action Fully Allowed
$R_b \to 0$ Braking action strongly penalized
Roadholding Objectives
$R_s \searrow$
$R_b \nearrow$
Adaptation to the driving Situations
More steering
Less Braking
Smooth transition between performance Objectives
$R_s = 1$ Steering action strongly penalized
$R_b \to 1$ Braking action Allowed
Comfort Objectives
Steering
Braking
Suspension
Emergency Level Of The driving Situation

Fig. 2. Actuator monitoring and scheduling strategy.

B. Classification of the Driving Situations

Based on the previously defined driving situation monitor R_b, the other varying parameter R_s allows to classify these driving situations depending on how dangerous and on the degree of emergency under which the vehicle is running. This parameter R_s is defined as follows:

$$R_s \begin{cases} = 1, & \text{when } 1 > R_b > R^2_{\text{crit}} \\ = \frac{R_b - R^1_{\text{crit}}}{R^2_{\text{crit}} - R^1_{\text{crit}}}, & \text{when } R^1_{\text{crit}} < R_b < R^2_{\text{crit}}. \\ = 0, & \text{when } 0 < R_b < R^1_{\text{crit}} \end{cases} \tag{2}$$

When $R_b > R^2_{\text{crit}} (= 0.9)$, i.e., when a low slip $(< s^-)$ is detected, the vehicle is not in an emergency situation, and R_s is set to 1. When $R_b < R^1_{\text{crit}} (= 0.7)$, i.e., when a high slip occurs $(> s^+)$, a critical situation is reached, and R_s is set to 0. Intermediate values of R_b correspond to intermediate driving situations.

The R_b and R_s varying parameters are used (as detailed later in the design step) to schedule the use of the active steering, semi-active suspension, and electromechanical braking actuators according to the driving situation and to optimize their operating range as described in the following and summarized in Fig. 2.

Normal situation: $(R_s = 1, R_b \to 1)$ The driving cruise goes smoothly (no emergency situations). Since there is no risk of wheel locking, as $R_b \to 0$, the rear braking torque is not limited. The semi-active suspension is tuned in order to preserve the comfort of passengers, without deteriorating the roadholding (i.e., soft suspension damping), due to the scheduling parameter $R_s = 1$. Moreover, since the driving situation is safe, no corrective steering action is needed to stabilize the vehicle.

Intermediate situation: $(R_s \searrow, R_b \searrow)$ As the driving situation becomes dangerous, the values of the scheduling parameters R_s and R_b change. The tire forces approach the nonlinear zone of the tire characteristic. As a result, the value of the monitor R_b starts to rise, and the braking torque is penalized to prevent the wheel locking. At the same time, the varying parameter R_s decreases, and a corrective steering action is allowed to help the driver to overcome this situation. Moreover, the semi-active suspension characteristics are changed smoothly, from soft to hard, depending on the value of R_s, to further improve the car roadholding without deteriorating the passengers comfort.

Critical situation: $(R_s = 0, R_b \to 0)$ When a dangerous situation is detected through the braking monitor $R_b = 0$ (in terms of longitudinal tire slip), the braking torque is limited accordingly in order to bring back the forces into the linear stable zone of the tire characteristic. As R_s reaches zero, the maximum additive steering angle is generated, and the semi-active MRDs are tuned to be "hard" to ensure a good roadholding (small wheel rebound). This will help the driver to overcome the critical driving situation and to prevent the vehicle from imminent accidents.

C. Global Chassis Controller Design Synthesis

The scheme in Fig. 1 shows the proposed two-step global chassis controller (GCC) LPV/$\mathcal{H}_\infty$ strategy. The first one is dedicated to the front steering/rear braking controller, which aims at improving the yaw stability and the lateral dynamics. The other one corresponds to the four semi-active MRD suspension systems, to enhance the vertical behavior (comfort/roadholding performances of the car). It is worth noting that the coupling effects are handled through the scheduling parameter R_s and due to an "antiroll" action of the semi-active suspension.

The main strategy is to adapt the control action to the driving situation, as presented in Fig. 2, using a self-scheduled controller function of R_b and R_s. This will be achieved through a good coordination and communication between the actuators of active front Steering, electromechanical rear braking, and the semi-active MRD suspension systems.

III. FIRST STEP: THE BRAKING/STEERING CONTROL PROBLEM FORMULATION

The LPV/$\mathcal{H}_\infty$ controller synthesis for the braking/steering subsystems is achieved, based on the extended bicycle model

$$\begin{bmatrix} \dot{v}_y \\ \dot{\psi} \\ \beta \end{bmatrix} = \begin{bmatrix} \frac{-C_f - C_r}{mv} & v - \frac{-C_f l_f + C_r l_r}{mv} & 0 \\ \frac{-C_f l_f + C_r l_r}{I_z v} & \frac{-C_f l_f^2 - C_r l_r^2}{I_z v} & 0 \\ 0 & 1 + \frac{l_r C_r - l_f C_f}{mv^2} & -\frac{C_f + C_r}{mv} \end{bmatrix} \begin{bmatrix} v_y \\ \psi \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{C_f}{m} & -\frac{1}{m} & 0 \\ \frac{C_f l_f}{I_z} & 0 & \frac{t_r}{R I_z} \\ \frac{C_f}{mv} & 0 & \frac{1}{mv} \end{bmatrix} \begin{bmatrix} \delta \\ F_{dy} \\ T_{b_{rj}} \end{bmatrix} \tag{3}$$

where $\delta = \delta^0 + \delta^+$ (the driver and **controller additive steering angle** input respectively), $\dot{v}_y$ is the lateral acceleration, v_y is vehicle lateral velocity, $\dot{\psi}$ is the yaw rate of the vehicle, β is the sideslip of the car, v is the vehicle speed, $C_{f,r}$ is the front and rear linear stiffness of the lateral tire, R is the nominal wheel radius, m is the vehicle mass, I_z is the yaw inertia, F_{dy} is the disturbance lateral force, $T_{b_{rj}}$ is the rear braking torque, and $t_{r,f}$ and l_{rf} are the (front, rear) axle and COG-front, rear) distance, respectively.

The model describes the lateral dynamics of the vehicle. It has been used in many studies to synthesize braking and steering control to enhance several dynamical behaviors such as the yaw rate, the lateral acceleration, and the lateral sideslip dynamics (more details see [12]).

The following general control configuration (including gain scheduled weighting functions) is considered.

$W_{e_{\dot\psi}} = 10((s/500+1)/(s/50+1))$ and is used to shape the yaw rate error ($e_{\dot\psi} = \dot\psi_{\text{ref}} - \dot\psi$), and $W_{\dot v_y} = 10^{-3}$ attenuates the lateral acceleration. Moreover, $W_{T_{b_{rj}}}(R_b) = (1-R_b)((s/10\varpi+1)/(s/100\varpi+1))$ attenuates the yaw moment control input according to the value of R_b, and $W_{\delta^+}(R_s) = R_s((s/\kappa+1)/(s/10\kappa+1))$ attenuates the steering control input according to the value of R_s, where ϖ (respectively, κ) is the braking (respectively, steering) actuator cutoff frequency.

The controller is chosen to be scheduled by the varying parameters R_b and R_s (according to the diagram given in Fig. 2) in order to achieve the following objectives.

- **Normal situation:** The tire force is in the linear friction zone, i.e., there is no risk of wheel locking; therefore, $R_b \to 1$ and the weighting function gain of $W_{T_{b_{rj}}}$ is chosen to be low. Therefore, the braking control is allowed to stabilize the vehicle. At the same time $R_s = 1$, the gain of the weighting function on the steering control is high, and no additive steering angle is necessary.
- **Intermediate situation:** When the driving situation becomes more dangerous, the gains of the weighting functions on the braking and steering actions change to cope with the needs for the vehicle stabilization. Indeed, the braking action is more and more reduced to avoid the wheels skidding, whereas a more corrective steering angle is supplied to help keeping the vehicle stable.
- **Critical situation:** When a high slip ratio is detected, $R_b \to 0$; the gain of the weighting function is increased to deactivate the braking torque and to prevent the wheels from locking. Then, the value of the varying parameter R_s is set to zero, the steering weighting function is no longer penalized, and a maximum corrective action by the steering actuators is allowed to compensate for the lack of braking and to preserve the handling and stability of the vehicle. This may help the driver to overcome the critical driving situations.

The corresponding LPV generalized plant $\Sigma(R(\cdot))$ is modeled as

$$\begin{bmatrix} \dot x \\ z \\ y \end{bmatrix} = \left[\begin{array}{c|cc} A(R_s,R_b) & B_1(R_s,R_b) & B_2 \\ \hline C_1(R_s,R_b) & D_{11}(R_s,R_b) & D_{12} \\ C_2 & 0 & 0 \end{array}\right] \left[\begin{array}{c} x \\ \hline w \\ u \end{array}\right] \tag{4}$$

where x includes the state variables of the system and of the weighing functions; $w = F_{dy}$ and $u = [\delta^+, T_{b_{rl}}, T_{b_{rr}}]$ are the exogenous and control inputs, respectively; $z = [W_{e_{\dot\psi}} e_{\dot\psi}, W_{\dot v_y} \dot v_y, W_{T_{b_{rj}}}(R_b) T_{b_{rj}}, W_{\delta^+}(R_s)\delta^+]$ holds for the controlled output; and $y = \dot\psi_{\text{ref}}(v) - \dot\psi$ is the controller input, where $\dot\psi_{\text{ref}}(v)$ is provided by a reference bicycle model. Notice that, the LPV model (4) is here affine with respect to the parameters R_s and R_b and can be described as a polytopic system, i.e., a convex combination of the systems defined at each vertex formed by $\mathcal{P}_R(\cdot)$, namely, $\Sigma(\underline{R(\cdot)})$ and $\Sigma(\overline{R(\cdot)})$. The controller is then a convex combination of four vertex controllers obtained at the min/max values of R_b/R_s. From the affine generalized plant in Fig. 3, an LPV polytopic controller is designed in the framework of the quadratic stabilization, as explained, for instance, in [15].

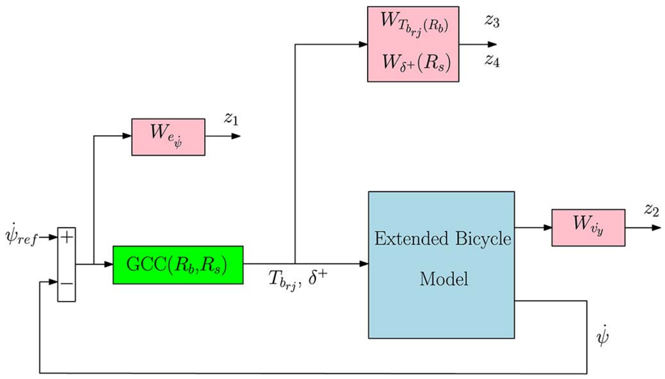

Fig. 3. Generalized plant for braking/steering control synthesis.

IV. SECOND STEP: THE SUSPENSION CONTROL PROBLEM FORMULATION

The control of the semi-active suspension system is synthesized using the classical seven-degrees-of-freedom vertical model (see [12]) in order to handle the tradeoff between the chassis motion (comfort) and the roll one (handling). Here, a vehicle equipped with four MR semi-active dampers is considered. As explained in Section IV-C, the MRD is a nonlinear component with dissipative capability used in automotive suspension control systems, where the damping property varies according to the applied magnetic field. Such a damper is able to provide adaptive performances in terms of comfort and roadholding. This model includes the vertical dynamics of the chassis z_s, the vertical motions of the wheels $z_{us_{ij}}$, the pitch ϕ, and the roll θ. The dynamical equations are

$$\begin{cases} \ddot z_s & = -(F_{sz_f} + F_{sz_r} + F_{dz})/m_s \\ \ddot z_{us_{ij}} & = (F_{sz_{ij}} - F_{tz_{ij}})/m_{us_{ij}} \\ \ddot\theta & = ((F_{sz_{rl}} - F_{sz_{rr}})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f)/I_x \\ \ddot\phi & = (F_{sz_f} l_f - F_{sz_r} l_r)/I_y \end{cases} \tag{5}$$

where F_{sz_i} is the vertical suspension forces; $F_{tz_{ij}}$ is the vertical tire forces; I_x, I_y, I_z is the roll, pitch, and yaw moments, respectively; F_{dz} is the disturbance yaw moment; and

$$\begin{cases} F_{tz_{ij}} = k_t(z_{us_{ij}} - z_{r_{ij}}) + c_t(\dot z_{us_{ij}} - \dot z_{r_{ij}}) \\ F_{sz_{ij}} v = F_k(z_{s_{ij}} - z_{us_{ij}}) + F_c(\dot z_{s_{ij}} - \dot z_{us_{ij}}) \text{ (passive)} \\ F_{sz_{ij}} = F_k(z_{s_{ij}} - z_{us_{ij}}) + u_{ij}^{\mathcal{H}_\infty} \text{ (controlled)} \end{cases} \tag{6}$$

where k_t and c_t are the linear tire stiffness and damping factors, respectively.

The control of suspension systems aims at enhancing the vertical dynamics of the vehicle in order to achieve frequency specification performances (see [7] and [16]). Here, the control objectives are oriented toward bounce and roll motions, characterized by the frequency-domain weighting functions in the $\mathcal{H}_\infty$ control framework (see Fig. 4), where $W_{z_s}(R_s) = R_s s^2 + 2\xi_{11}/\Omega_{11} s + {\Omega_{11}}^2 s^2 + 2\xi_{12}\Omega_{12} s + {\Omega_{12}}^2$ is shaped in order to reduce the bounce amplification of the suspended mass (z_s) between [0,12] Hz, when R_s is high.

$W_\theta(R_s) = (1-R_s)s^2 + 2\xi_{21}/\Omega_{21} s + {\Omega_{21}}^2 s^2 + 2\xi_{22}\Omega_{22} s + {\Omega_{22}}^2$ attenuates the roll amplification in low frequencies when R_s is low.

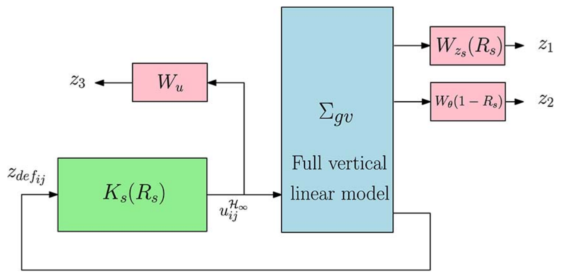

Fig. 4. Suspension system generalized plant.

$W_u = 3.10^{-2}$ is set to shape the control signal.

This control design schedules the use of the semi-active dampers and the vehicle performance as follows:

Normal situation: $R_s = 1$, and the semi-active suspension control enhances the passenger comfort objectives by using a high gain of the weighting function on the chassis displacement z_s. The undesirable vibrations of the chassis are then absorbed by the semi-active MRDs, which are tuned to have a soft damping characteristic.

Intermediate situation: When the driving situation changes, the varying parameter R_s decreases, which increases the weighting on the roll dynamics of the car caused by the lateral load transfers. Therefore, the suspension control modifies the performance objectives from passenger comfort to roadholding. The LPV framework used in the proposed strategy ensures a smooth and efficient transition between these performance objectives while ensuring the stability conditions.

Critical situation: $R_s = 0$, and the semi-active suspension control acts to further improve the roadholding. The weighting on the chassis motion is relaxed since the passenger comfort is no longer the priority, and a high penalization on the roll motion is set to reduce the load transfer that may lead to vehicle instability (close to accident).

According to Fig. 4, the following parameter-dependent generalized plant $(\Sigma_{gv}(R_s))$ is obtained:

$$\begin{bmatrix} \dot{\xi} \\ \hline z \\ y \end{bmatrix} = \left[\begin{array}{c|cc} A(R_s) & B_1(R_s) & B_2 \\ \hline C_1(R_s) & D_{11}(R_s) & D_{12} \\ C_2 & 0 & 0 \end{array}\right] \begin{bmatrix} \xi \\ \hline w \\ u \end{bmatrix} \quad (7)$$

where $\xi = [\chi_{vert}\ \chi_w]^T$ is the state vector of the system plus the state vector of the weighting functions; $\tilde{z} = [z_1\ z_2\ z_3]^T$; $\tilde{w} = [z_{rij}\ F_{dx,y,z}\ M_{dx,y}]^T$; $y = z_{\mathrm{def}_{ij}}$; $u = u_{ij}^{\mathcal{H}_\infty}$, with $i = f, r$ and $j = l, r$; F_{dz} is the vertical disturbance; and M_{dz} is the disturbance moment along the z-axis.

The LPV system (7) includes a single scheduling parameter (R_s) and can be described as a polytopic system after some relaxation, i.e., a convex combination of the systems defined at each vertex of a polytope defined by the bounds of the varying parameter. The LPV/$\mathcal{H}_\infty$ suspension controller synthesis is obtained due to LMI's resolution of the control problem following the mathematical development given in [17].

Remark 1: Since semi-active suspensions are considered, the LPV controllers are clipped in order to cope with the damper constraints.

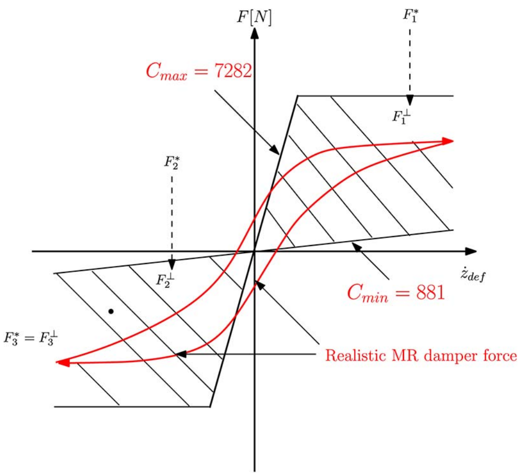

Fig. 5. Projection principle of the semi-active controlled damper model (F_1^* and F_2^* are out of the allowed area and F_3^* is inside) + the MRD force with biviscosity "$C_{\min} = 881, C_{\max} = 7282$" (for more details, see [19]).

A. *Semi-Active Suspension Control Implementation*

The application of the proposed LPV control to the considered semi-active suspension is achieved here, for simplicity, using the clipped strategy (see [7]). Fig. 5 shows the experimental characteristics of the MRDs obtained in collaboration with colleagues from ITESM, Monterrey, Mexico (see [18]). Given a deflection speed $\dot{z}_{\mathrm{def}}$ and a desired controlled damper force F^*, the clipped approach consists in projecting F^* onto the admissible force domain, if necessary, to get $F^\perp$.

V. SIMULATIONS

The nonlinear model used for the simulations purposes was validated by an experimental procedure made on a real Renault Mégane Coupé, through a Moose test performed on a real track (see [12]). Then, two different simulation scenarios are given, and the corresponding results are analyzed. The first scenario is a double-line change simulation based on the validated nonlinear model, and the second one is based on the experimental data given by a performed moose test on a real race track.

A. *Simulation Results: A First Scenario*

In this case, simulation results are presented to emphasize the improvements of LPV closed-loop control (denoted "CL LPV semi-active") compared to the open-loop results (denoted "Open Loop") and, for sake of completeness, to the standard LTI/$\mathcal{H}_\infty$ design of both active steering/braking controller and semi-active suspension controller (without scheduled gains), (denoted "CL LTI", which was achieved by solving the previous $\mathcal{H}_\infty$ problems with the values of the varying parameters frozen at $R_s = 0.1$ and $R_b = 0.1$. The following scenario is considered. When the vehicle runs at 100 km/h in straight line, the following events occur. From $t = 0.5$ s to $t = 1$ s, there was a 5-cm bump on the left wheels. Then, the driver performed a double line change from $t = 2$ s to $t = 6$ s. Finally, there was another 5 cm bump on the left wheels, during the maneuver,

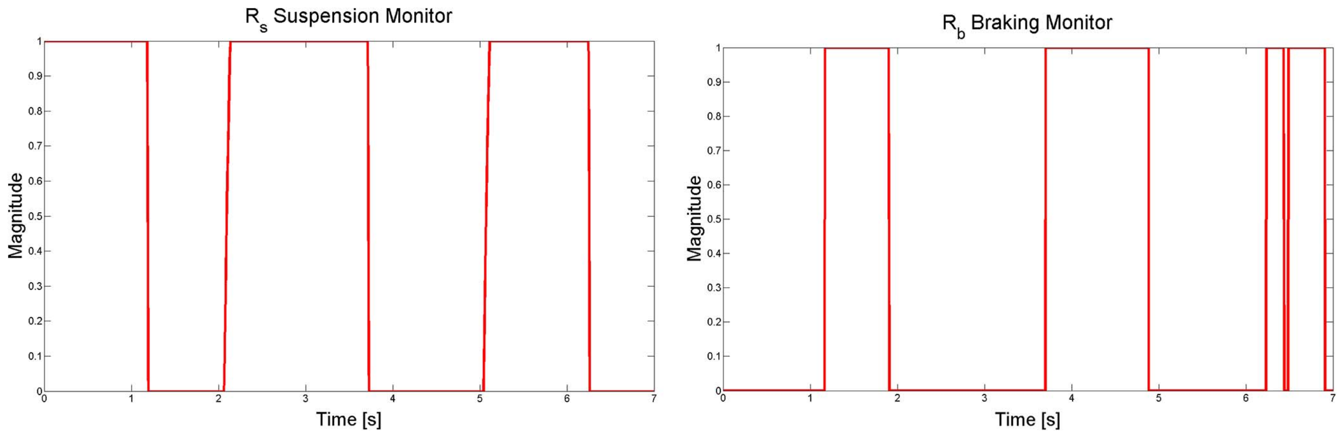

Fig. 6. Monitoring signals.

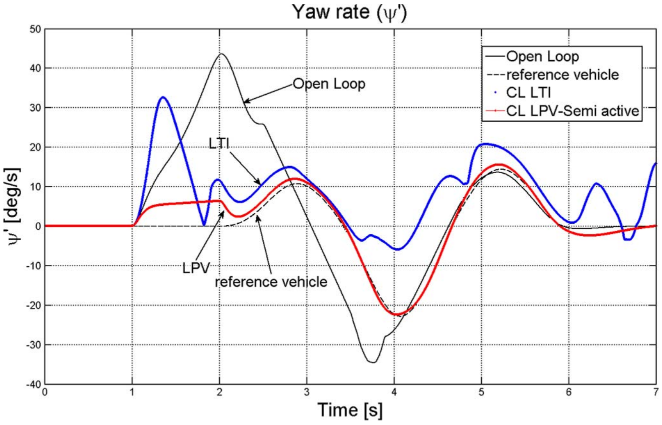

Fig. 7. Yaw rate.

from $t = 3$ s to $t = 3.5$ s. A lateral wind that occurs at vehicle's front, generating an undesirable yaw moment, is considered ($t = 2.5$ s to $t = 3$ s). In this scenario, for the robustness analysis, the road is considered wet ($\mu = 0.5$, the road adherence parameter), which reduces the road/tire adhesion and the lateral tire contact forces.

The resulting monitoring signals R_b [see (1)] and R_s [see (2)] are shown in Fig. 6 and justify the LPV framework of the strategy. The varying parameters R_b and R_s allow to activate, limit, or deactivate the control action when required (for braking and steering actuators). Let us recall that the R_s scheduling parameter depends on the value of R_b, which itself depends on the slip ratio dynamics. These parameters are very important since they define the behavior of the vehicle subject to critical driving situations. They will be used to provide the driver with the necessary assistance, through the steering, braking, and suspension subsystems.

1) Lateral Dynamics Behavior Analysis: It is shown in Fig. 7 that the proposed LPV/$\mathcal{H}_\infty$ strategy enhances better the lateral dynamics (here, the vehicle yaw tracking). Compared with the LTI/$\mathcal{H}_\infty$ controller, it gives good results in terms of vehicle lateral stability.

Remark 2: Simulations using an extended bicycle model with the driver input have given the "ideal" reference vehicle to be tracked by the vehicle (black dashed line, see Fig. 7). It helps to compare and to emphasize the improvements brought by the proposed LPV/$\mathcal{H}_\infty$ strategy.

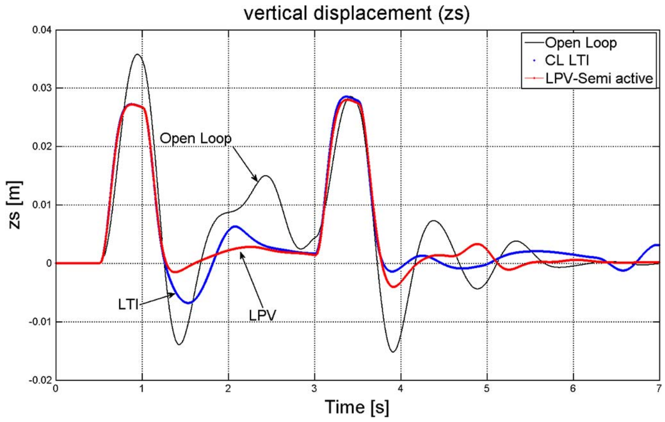

Fig. 8. Vertical chassis displacement z_s.

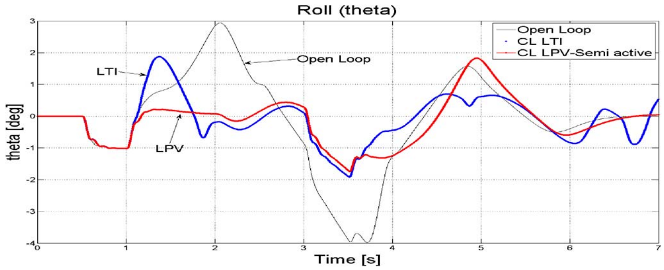

Fig. 9. Roll motion of the chassis θ.

2) Vertical Dynamics Behavior Analysis: The vertical motion of the chassis is shown in Fig. 8. The LPV/$\mathcal{H}_\infty$ controller improves the vertical dynamics better than the LTI/$\mathcal{H}_\infty$ one does. The chassis displacement is considerably reduced by the proposed strategy. This enhances the passengers comfort while driving on uneven roads.

Fig. 9 represents the improvement brought in term of the load transfer mitigation. The roll motion is well attenuated which, in addition to enhance the vehicle stability, ensures a good road handling of the vehicle running in dangerous driving situations. It is seen that the use of the semi-active suspension

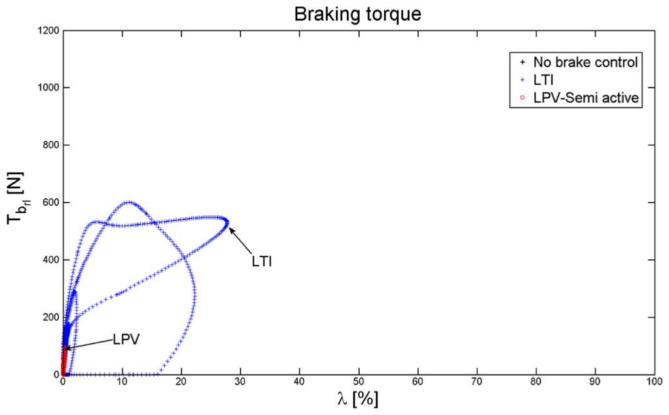

Fig. 10. Rear right breaking torque.

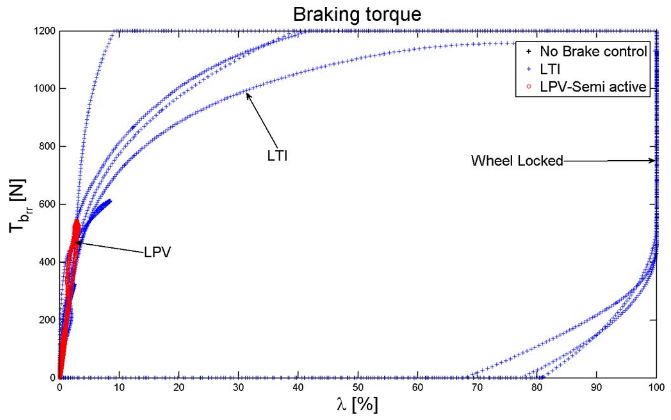

Fig. 11. Rear left breaking torque.

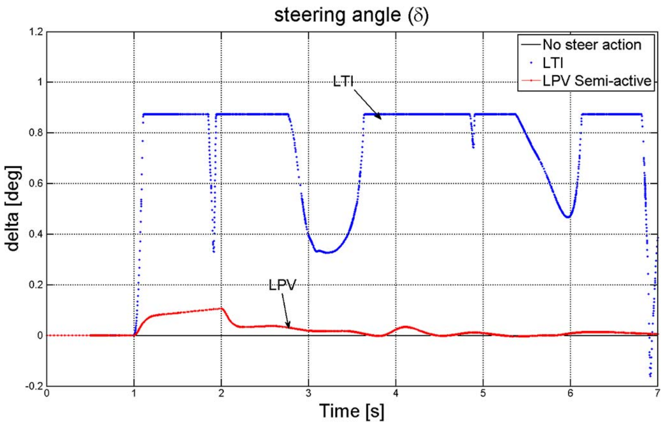

Fig. 12. Steer control input.

TABLE I
PERFORMANCES EVALUATION

Signals	improvement %	Vehicle dynamics
zs	11	Chassis displacement
θ	13	Roll motion
$\dot{\psi}$	19	Yaw rate
$\dot{y}$	32	Lateral acceleration
$T_{b_{rl}}; T_{b_{rr}}$	68; 74	Braking torques
δ	86;	Steering angle
Fs_{ij}	27, 31, 23; 28	Suspension forces

control in the coordinated "LPV/$\mathcal{H}_\infty$" strategy (with hierarchical activation of the different actuators depending on the driving situations needs) gives better results than in the "LTI" case.

3) Actuators Dynamics Behavior Analysis: In addition to enhancing the vehicle various dynamics, the proposed LPV/$\mathcal{H}_\infty$ improves the use of the actuators (electromechanical braking, active steering, and semi-active suspensions) considered for the car under study. The following figures show interesting results for the actuator activation. Figs. 10 and 11 show the braking torque provided by the vehicle to perform the previously defined scenario. The braking torque provided by the LPV/$\mathcal{H}_\infty$ controller is depicted in red. The torque is clearly much lower than that provided in the LTI controller case (blue curves), which saturates. Moreover, the use of the LPV/$\mathcal{H}_\infty$ strategy avoids wheel locking. Fig. 11 shows that, for the LTI case, the longitudinal slip ratio λ_{rl} reaches the 100% value, which means that the left rear wheel is locked. Fig. 12 shows also that the "LPV coordination strategy" can help the driver to keep the vehicle stable with a minimum effort. The steer control considerably decreases in the "LPV" case compared with the "LTI" case and is activated only when the driving situation is dangerous enough.

Therefore, the "LPV coordination strategy" can help the driver to keep the vehicle stable with a minimum effort. Indeed the steer control considerably decreases in the "LPV" case, compared with the "LTI" case, and is activated only when the driving situation is dangerous enough.

Remark 3: In the previous simulations, the LTI control strategy gives good results. However, since the rear wheels lock during the maneuver (see Fig. 11), leading to a very high risk of loss of maneuverability and safety degradation, the LPV control appears to be a very efficient way to deal with the braking issues. Moreover, it enhances performances and stability, using the previously presented integrated control strategy. Furthermore, the LPV controller uses the actuators of braking, steering, and suspension in a coordinated way to enhance the overall vehicle dynamics and to cope better with the actuators characteristics and limitations.

The improvements brought by the proposed strategy compared with the LTI control case are quantified in Table I by calculating the real mean square values of the different car dynamics signals. The comparison shown in Table I proves the efficiency of the proposed solution for this driving scenario.

B. Simulation Results: A Second Scenario

This scenario uses experimental data obtained for model identification. Indeed, a test (of the real uncontrolled Renault Mégane Coupé car) was first performed by a professional driver on a real race track. This circuit includes a left bend and then an obstacle avoidance (emergency situation) to determine how well a vehicle evades a suddenly appearing obstacle.

In the considered simulation case, the focus is put on the "Moose" test only (performed at a velocity of 90 km/h^{-1}) in order to assess the efficiency of the designed controllers for obstacle avoidance.

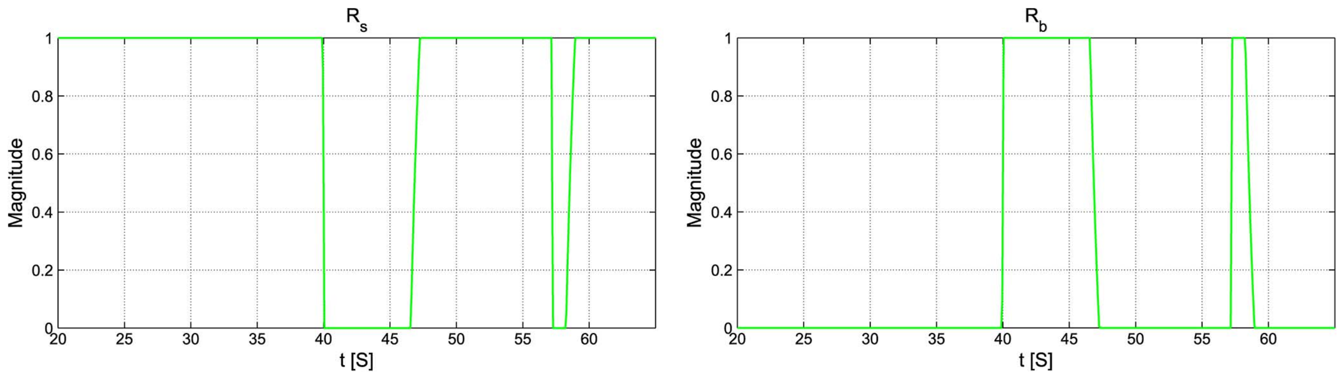

Fig. 13. Monitoring R_s and R_b signals.

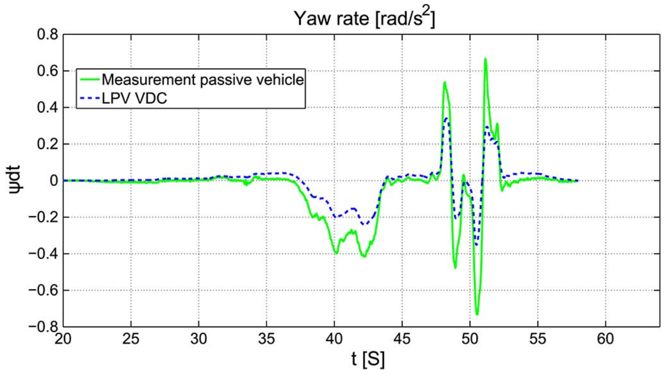

Fig. 14. Yaw rate $\dot{\psi}$.

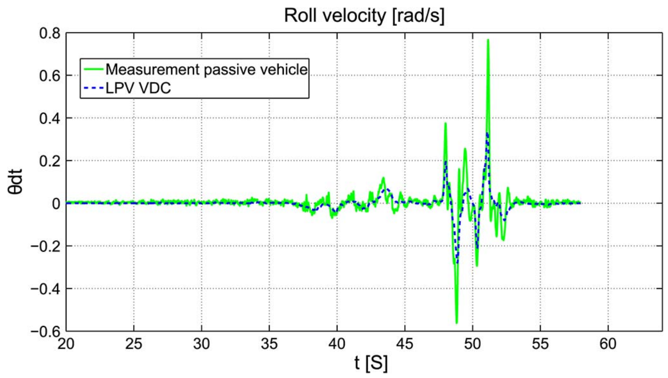

Fig. 15. Roll velocity of the chassis $\dot{\theta}$.

The "driver" inputs (i.e., the steering angle and the longitudinal speed) are considered external inputs of the NL closed-loop model for the simulation of the LPV control. The closed-loop simulation results obtained from real input data (denoted here as "LPV VDC") are then compared with the experimental ones (denoted here as "Measurement passive vehicle").

The resulting varying parameters R_s and R_b, which schedule the coordination of the three actuator's controllers (semi-active suspension, active steering, and electromechanical braking) are shown in Fig. 13. These parameters have complementary values, which is coherent with the previously presented monitoring strategy.

Remark 4: The passive vehicle dynamical behaviors presented in this section were measured on the real vehicle (Renault Mégane Coupé) while performing this scenario on a real circuit path.

Fig. 14 shows the yaw rate behavior of the vehicle using the proposed LPV/$\mathcal{H}_\infty$ control compared to the passive vehicle behavior. One can notice that the yaw rate dynamics of the vehicle are well improved even if the vehicle is running with a quite high velocity (90 km/h^{-1}) on the left bend when avoiding the obstacle.

Fig. 15 shows the improvement of the roll velocity. Indeed, using the designed LPV/$\mathcal{H}_\infty$ controllers that coordinate the use of the semi-active suspension, steering, and braking, the roll motion considerably reduces (47% less than that of the passive car). It is obvious that the vertical dynamics are better enhanced using an LPV/$\mathcal{H}_\infty$ robust controller in emergency situations.

Fig. 16 shows the measured rotation angle (left) of the steering wheel that the driver generates to perform the considered driving scenario and the corrective steering angle (right) that the LPV controller supplies to help the driver to ensure the vehicle stability and maneuverability. This corrective steering angle is directly applied on the wheel (not on the steering wheel).

Notice that the steering ratio of the car, which is the rotation angle of a steering wheel divided by the steer angle of the wheels, is around 10:1 to 20:1, depending on the car's type (commercial, race, sport, etc.). This means that the corrective steering angle's effect is very important.

VI. CONCLUSION

In this paper, a global chassis control strategy has been proposed, involving active steering, electromechanical braking, and semi-active suspension. This strategy was shown to enhance the vehicle dynamical behavior subject to critical driving situations. In this framework, the LPV approach plays a major role to efficiently schedule the use of these actuators. Indeed, the originality of the proposed approach is first concerned by the coordinated use of these three types of actuators and second by their hierarchical activation, depending on the driving situations, which allows to reach the performance objectives.

Another advantage of the LPV methodology (compared with classical LTI controllers) is the limitation of the braking actuation in critical situations to avoid wheel locking and skidding, and its coordination with active steering and semi-active

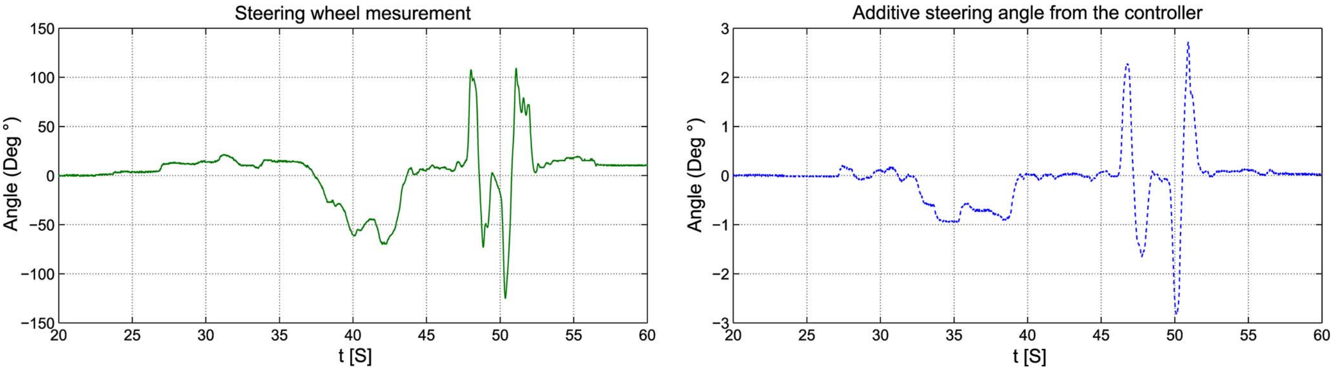

Fig. 16. (Left) Steering wheel angle δ^0 and (right) corrective steering angle from the controller δ^+.

suspension controllers, leading to vehicle stability and road handling improvements.

Simulation results, obtained from experimental input data and performed with a validated complex nonlinear vehicle model, have assessed the performances of the proposed approach. However, a complete control validation step requires a set of experiments performed on a test car equipped with the considered actuators. The real implementation of the control algorithm might lead to several problems that do not occur in simulation, for instance, real-time constraints. This could be handled further in an experimental study.

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